Topology optimization of an automotive hood for multiple load cases and disciplines

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1 Abstract

To reduce the head impact injuries in case of traffic accidents, the design of an automotive hood must consider many design requirements including impact of the head against the hood at different locations, be lightweight but with enough stiffness to resist various loads imposed on the hood, and have NVH characteristics such as the fundamental frequency. Methodologies to solve this type of design optimization problem that integrates multiple design criteria are rare to non-existent in the automotive design field. This paper shows how to conduct the worst-case design of the hood for multiple head impact locations, which is required by the pedestrian safety code. In addition, a topology optimization problem of the hood that combines statics, impact, and eigen frequency load cases is solved by using LS-TaSC to provide the optimal lightweight hood structure satisfying the design constraints. This is possibly the first demonstration of both the worst-case design and multi-disciplinary design optimization considering both impact and frequency load cases on an industrial problem.

Keywords: Automotive Hood; Head Injury Criterion; Multi-disciplinary Design; Worst-case Design;

2 Introduction

Topology optimization finds the layout of structure to support a specific set of loads. It is preferably used in the initial stages of structural designs. Historically the methods have been developed for only a single load, but the adoption of the method requires that multiple loads from different disciplines be considered in addition to design codes.

The automotive front-end designs for crash box and engine hood are of great importance to protect the automobiles from serious damage and reduce injuries to the pedestrians at the time of collisions. Designing the automotive crash box and hood involves many design requirements including strength, stiffness, and frequency characteristics. For example, the design of an automotive crash box can focus on seeking for the lightest structure with improved crashworthiness performances [1-2], in terms of energy absorption abilities, damage level to the driver, and required natural frequencies.

The design of the hood of a car is more complex than that would appear from initial inspection. One would expect static loads and a fundamental frequency consideration. But it ends up that a major consideration is the pedestrian impact which is regulated by Euro new car assessment program (NCAP) Pedestrian testing protocol. Because the automotive runs under the pedestrian and the severity of injuries vastly depend on the vehicle front shape and certain characteristics such as energy absorption [3]. The Head Injury Criterion (HIC) is used as a major indicator of measuring the head injury during a pedestrian impact by the Euro NCAP test protocol. Thus, the structural design of an automotive hood is made very complex through the pedestrian safety protection requirement.

Relevant research on the structural design of an engine hood has been conducted over the past few years. Salway and Zeguer investigated hood design in [4] in which they considered both topology and parametric optimization, but multi-disciplinary optimization was only possible for parametric optimization at the time. Valeria and coworkers [5] presented the topological and topographic optimization of a hood for lightweight structure with target stiffness and frequency characteristics by using OptiStruct. Tang and coworkers [6] studied the topology optimization of inner panel of a SUV hood by using OptiStruct as well. Li and Hu et al. [7] conducted topology optimization design research for aluminum inner panel of engine hood and mucilage glue regions. All these researchers obtained some good conceptual designs of the hood structure under multiple load cases, though limited with regard to the static and NVH performances of the engine hood. There is a missing consideration for the pedestrian safety protection requirement in their designs. Hence, it is of great significance to propose a methodology that can address topology optimization design of the engine hood performing under statics, pedestrian impact, and NVH load cases simutaneously. In this work we show a higher maturity level of topology optimization able to handle multidisciplinary optimization amongst others.

LS-TaSC has been developed by Livermore Software Technology, an ANSYS company, initially for highly nonlinear mechanics, but has been expanded for the whole range of automobile design. The project introduced innovations such as multi-point methods for constrained topology of impact problems; the theory behind maximization of energy absorption for impact problems; multi-disciplinary topology optimization involving impact, statics, and NVH; as well as worst-case topology optimization.

3 Multidisciplinary Topology Optimization with Constraints

To solve for Multidisciplinary Design Optimization (MDO) problems, we introduce some additional variables $\boldsymbol{\xi}$ (known as the spatial kernel), in additional to the normal topology variables $\boldsymbol{x}$.

A generalized problem can be solved considering the dual problem:

$$min_{\xi} F(\boldsymbol{\xi}(\boldsymbol{x})) \tag{1}$$

where $\boldsymbol{x}$ is computed using

$$min_{x} f(\boldsymbol{x}(\boldsymbol{\xi})) \tag{2}$$

with f usually taken as the compliance, or the negative value of fundamental frequency, which means the analyst only must specify F. In such a case one can maximize energy absorption using F while maximize stiffness using f, or you can minimize mass using F while maximizing stiffness using f.

To set it up as a constrained problem we add the constraints as

$$g_i(\boldsymbol{x}) \leq 0 \tag{3}$$

The constraints can be split into two sets -- for the one set design sensitivity information can be analytically computed

$$g_i^{ana}(\boldsymbol{x}) \leq 0 \ with\, i = 1, \dots, n \tag{4}$$

while the other set requires the computation of numerical derivatives using the spatial kernel in the upper problem

$$g_j^{num}(\boldsymbol{x}) \leq 0 \ with\, j = 1, \dots, m \tag{5}$$

Adding the Lagrange multipliers to the objective gives us the Lagrange function

$$L(\boldsymbol{x}, \boldsymbol{\lambda}, \boldsymbol{\xi}) = f(\boldsymbol{x}) + \sum_i \lambda_i g_i^{ana}(\boldsymbol{x}) + \sum_j \xi_j g_j^{num}(\boldsymbol{x}) \tag{6}$$

The constraints needing numerical derivatives are given special treatment. The spatial kernel

$$s(\boldsymbol{\xi}) = \sum_j \xi_j S_j(\boldsymbol{\zeta}) \tag{7}$$

is introduced to satisfy these constraints. The kernel is composed of basis functions referring to $\boldsymbol{\zeta}$ the spatial coordinates associated with variable $\boldsymbol{x}$. In our current implementation of the spatial kernel we roll up all the spatial kernel functions into a surface written as a summation over both the basis functions and the elements as:

$$h(\boldsymbol{x}) = \frac{1}{n} \sum_{e=1}^{N} \frac{x_e}{exp(\xi_0 + \xi_1 S_1(\zeta_e) + \xi_2 S_2(\zeta_e) + \cdots)} = 1 \tag{8}$$

with x_e a spatial value at element e $(e = 1, \dots, N)$, and $\boldsymbol{\xi}$ solved to satisfy the constraints. See reference [8] for the details of an implementation.

For multidisciplinary optimization we have the objective as

$$f(\boldsymbol{x}) = \sum_{lc} w_{lc} f_{lc}(\boldsymbol{x}) \tag{9}$$

in which it should be noted that the load case weights can be used to solve for a subset of the constraints. The constraints are split into two sets as described before -- for the one set design sensitivity information can be computed, while the other set requires the computation of numerical derivatives using the spatial kernel in the upper problem.

We can add the Lagrange multipliers to the objective giving the Lagrange function

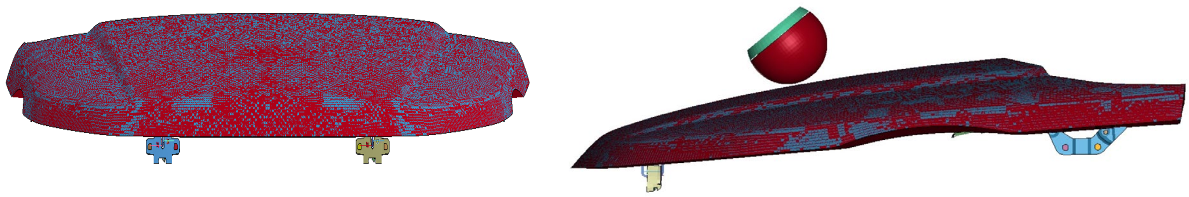
$$L(\boldsymbol{x}, \boldsymbol{w}, \boldsymbol{\lambda}, \boldsymbol{\xi}) = \sum_{lc} w_{lc} [1 + s(\boldsymbol{\xi})] f_{lc}(\boldsymbol{x}) + \sum_i \lambda_i g_i^{ana}(\boldsymbol{x}) \tag{10}$$

which contains the high-level variables $[\boldsymbol{w}, \boldsymbol{\lambda}, \boldsymbol{\xi}]$ used to solve for the constraints.

The dual problem is solved as an upper level problem in the Lagrange multipliers (including weight and spatial kernel variables) and a lower level problem in the topology variables. The lower level problem is solved using the projected subgradient method [9] considering the Lagrange multipliers, while the upper level problem can be solved using finite differences or surrogate models.

4 Designing an Engine Hood

The geometry of the engine hood includes the outer shell and the solid inner panel. The solid inner panel is the design area for topology optimization, and the finite element model of the whole engine hood including the outer shell is shown in Fig. 1(a). The head impact test included in the Euro NCAP safety assessment is particularly designed for evaluating the head injury arising from a pedestrian impact. Thus, a finite element model of child/small adult headform, weighing 3.5 kg, was positioned over the outer shell surface of the engine hood, as shown in Fig. 1(b). The baseline head impact analysis, which is to study the response of the headform collision on the engine hood, as well as the static loading analysis and the NVH analysis are conducted using LS-DYNA.



(a) FE model of the engine hood (b) FE model of the headform over the engine hood

Fig. 1: FE model of the engine hood for head impact analysis

4.1 Requirements for Pedestrian Safety Protection

The HIC value in a head impact analysis is calculated by the effects of head form acceleration and the duration of the acceleration. It is defined as

$$HIC = max\left\{\left[\frac{1}{t_2-t_1}\int_{t_1}^{t_2} a(t)dt\right]^{2.5}(t_2-t_1)\right\} \tag{11}$$

where $a(t)$ indicates the acceleration of the head form at time t, and t_1 and t_2 represent the time interval for the evaluation. The index 2.5 is chosen for the head, based on experiments. The HIC value is to evaluate the maximum average acceleration value over time duration $t_2 - t_1$. For the results presented in this article, an HIC time interval of 15ms was used. Generally, experts agree that HIC values above 1000 are life threatening [10]. Thus, a threshold of 1000 is set as the upper bound of the HIC values in all the design studies in this article.

According to the Euro NCAP test protocol, areas of the engine hood need to be defined for head impact positions for the design to meet low speed requirement, as shown in Fig. 2. It is required that the engine hood design must be evaluated at multiple impact locations in Test zone I and zone II. Ideally, all head impact points should be considered in a multi-load case topology optimization problem to obtain a practical design of the engine hood, but it is impossible to do so due to the enormous computation costs. It is more practical and conductible for topology optimization design to consider only one head impact point at one time of the design process. To obtain a practical design, a detailed computation strategy of the worst-case design will be proposed in Section 5.

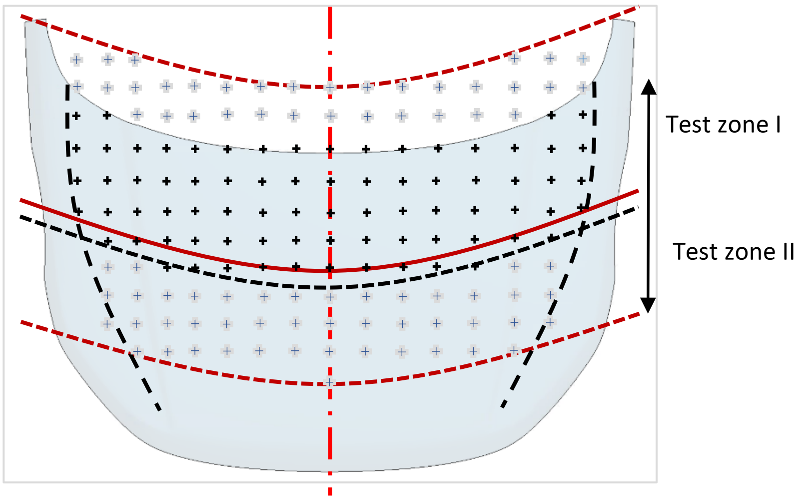


Fig. 2: Zoning of pedestrian head impact protection for Euro NCAP protocol

5 Automotive Hood Topology Optimization Design Studies

The goal of topology optimization of the automotive hood is to minimize mass while maintaining stiffness and frequency characteristics, as well as satisfying the pedestrian safety requirement. Five design studies have been conducted on the topology optimization of engine hood for investigating the capabilities of LS-TaSC in solving multidisciplinary design optimization problems.

5.1 Design Study 1: Design for Three Static Load Cases

Three static stiffness load cases of the automotive hood are considered in this study, including rear beam stiffness, torsional stiffness, and bending stiffness, as shown in Fig. 3. The optimization problem is formulated to minimize mass with respect to three constraints on the rear beam, bending, and torsion displacements. The optimized structure of the inner panel of the hood structure is shown in Fig. 4(a), whereas Fig. 4(b) shows the design contribution plot of three static load cases, within which the parts of the structure in red colour contribute to the bending load case, the parts in green colour contribute to the torsion load case, and the parts in blue colour contribute to the rear beam load case.

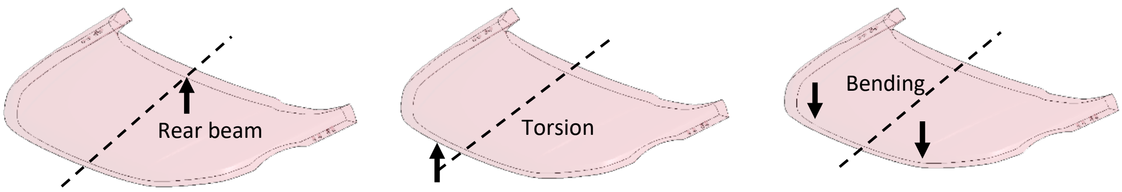
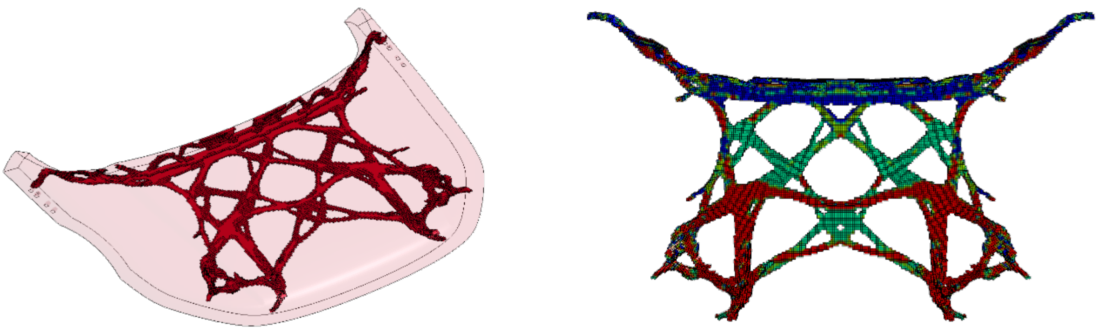


Fig. 3: FE model of the engine hood and three static load cases



(a) Optimum inner panel structure (b) Design contribution plot

Fig. 4: Optimized structure of the inner panel of the engine hood for three static load cases

5.2 Design Study 2: Design for Impact Load Case

A single head impact load case is considered in the design study to investigate how well the HIC requirement being imposed on the design. The child/small adult headform model is impacted against the vehicle hood at a velocity of 25 mph. The optimization problem is formulated to minimize mass with respect to a constraint on HIC value and a constraint on the displacement at the impact location. It is required that the displacement of the hood at impact location should be no more than 70 mm. This upper limit on the hood displacement was considered to compensate for the missing hard parts, such as engine, under the hood that would result in a high HIC value if the structure is too soft. The optimized structure of the inner panel is shown in Fig. 5.

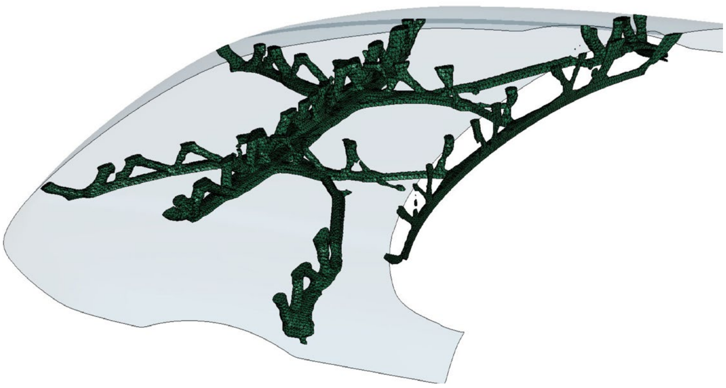


Fig. 5: Optimized structure of the inner panel of the engine hood for a single impact load case

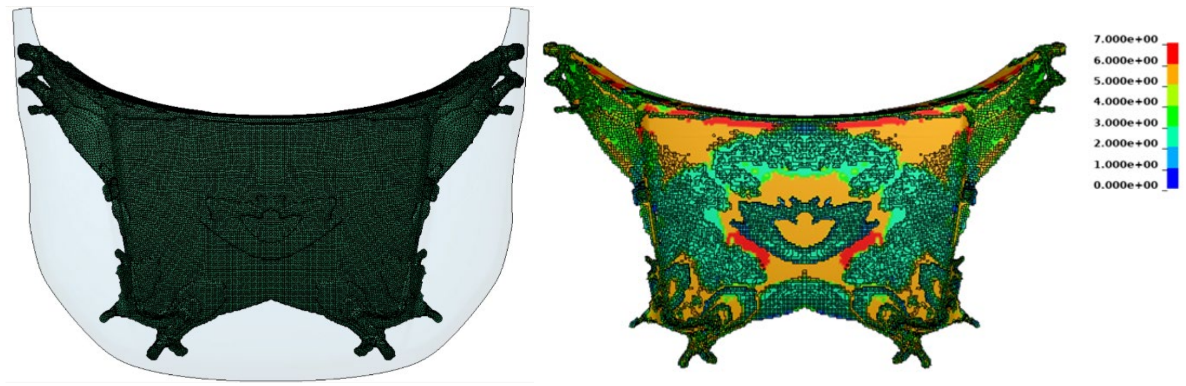
5.3 Design Study 3: Design for Impact and Static Load Case

This study is a design to investigate the dominancy of the head impact analysis over the other analysis. Thus, the optimization problem is formulated to minimize mass with respect to a HIC constraint and a torsion displacement constraint. Therefore, it is multidisciplinary optimization problem involving both impact and static load cases. It is required that the torsion displacement at the static loading point is no more than 17.6 mm. The final design of the inner panel of the hood is shown in Fig. 6. Both the HIC and torsion displacement constraints are satisfied at the final design.

Fig. 6: Optimized structure of the inner panel of the engine hood for impact and static load cases

5.4 Design Study 4: Design for Impact, NVH, and Static Load Cases

A more complex study is conducted to consider three load cases simultaneously, including a head impact load case, a torsional displacement analysis, and a modal frequency analysis. The optimization problem is formulated to minimize mass with respect to a HIC value constraint, a constraint on maximum downward displacement of the hood to be less than 100 mm, a torsion displacement constraint, and a bending frequency constraint. The bending frequency of the design is required to be no less than 45 Hz. The optimized final structure is shown in Fig. 7(a), and the design contribution plot is shown in Fig. 7(b), where the parts in red colour contribute to all three load cases, the parts in yellow colour contribute to the static and NVH load cases, and the parts in green colour contribute to the NVH load case only.



(a) Final design of the inner panel (b) Design contribution plot

Fig. 7: Optimized structure of the inner panel for three load cases

5.5 Design Study 5: Design for Multiple Impact Locations

As aformentioned in the pedestrian safety protection requirement, it is extremely costy to consider multiple impact locations for topology optimization. For example, a multi-load case topology optimization problem considering five different head impact locations may require approximately 30 LS-DYNA runs for each LS-TaSC iteration. Assuming convergence at 25 to 30 topology optimization iterations, the overall computation cost could be as high as 900 LS-DYNA runs. In order to acheive a practical design in a reasonal number of runs, we propose a computation strategy to achieve the design goal of multiple impact locations.

At first, according to the test zones definition, a total of 13 impact locations of interest are selected as design load cases and topology optimization is conducted for each impact location individually. Therefore, 13 individual topology optimization runs are conducted using LS-TaSC. This process can be automated using a parameteric optimization tool, such as LS-OPT, to parameterize and sample the impact locations and call LS-TaSC as a solver for topology optimization. The overylay of resulting 13 optimum topologies is shown in Fig. 8, where the color at each impact location corresponds to the color bar representing the HIC values obtained at that impact location.

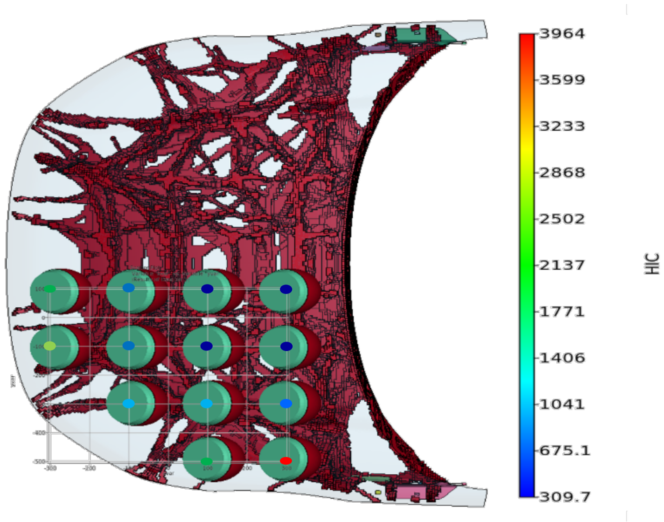


Fig. 8: Overlay of 13 topology optimization designs from 13 single load cases

The purpose of this study is to draw a conceptual idea of the potential load path from multiple impact locations. The results indicates that the impact location at the bottom-right corner where a high HIC value occures is the worst impact location among all of the locations.

Furthermore, topology optimization of the hood is conducted using worst-case head impact location and point loads applied at several other locations on the outer surface of the hood, as shown in Fig. 9.

Fig. 9: Load cases used for worst-case topology optimization

The upper bound of the HIC constraint in this worst case design is set as 2000 as the impact location is on the outer region of the hood, compared to the HIC requirement of less than 1000 for inner region. In this study, the goal of optimization is to find a reasonable load bearing structure based on the point loads and HIC requirement is taken into consideration using the impact load case. Therefore, a load case weight ratio of 99:1 is introduced instead of using equal weights or treating weights as additional parameters. The resulting optimum topology is shown in Fig. 10.

Fig. 10: Optimized structure of the hood for combination of worst impact location and point loads

To verify the proposed worst-case topology optimization approach, the locations of point loads where replaced with headform impact model and LS-DYNA analysis of the optimum topology is conducted to check the resulting HIC values and maximum downward displacement of the hood. Fig. 11 shows the impact locations and corresponding HIC values.

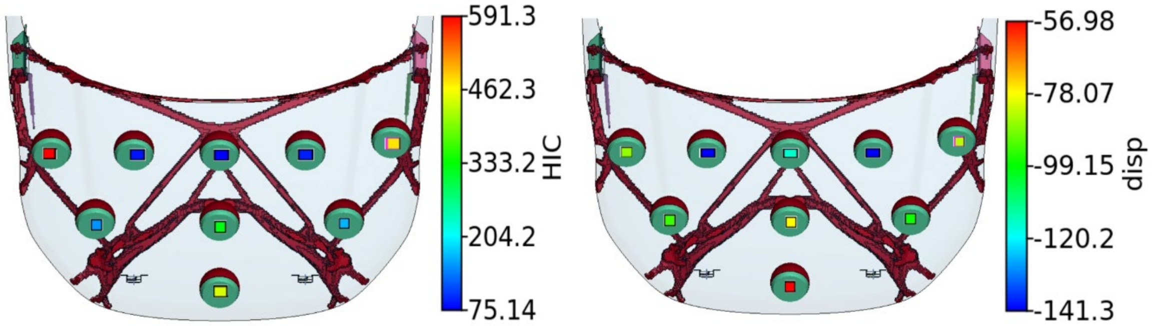


Fig. 11: HIC (left) and displacement (right) values for different impact locations. Note that the locations and therefore the results are not symmetric.

In above figure, the HIC values of all the impact location is within the required limit. However, for some of the impact locations, especially at top center locations, the maximium downward displacement of the hood has exceeded the limit of 100 mm. Therefore, considering top center impact location as additional impact load case for worst-case design may help satisfy this constraint; or in more general terms: worst case design of highly nonlinear problems seems to require an iterative selection of the worst case(s), a statement which needs further work to confirm.

6 Summary

The work demonstrated a new maturity level for topology optimization. The hood was designed for multiple, multi-disciplinary load cases including complex constraints such as the head impact criterion from the pedestrian impact design code. Additionally, we considered worst-case design, which was most likely the first demonstration of worst-case design in topology optimization. The multidisciplinary design of such structures is of the essence – one should not design a hood for the statics cases alone, in fact it is a regulatory requirement that it be design for head impact as well.

7 Literature

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